Glass and Ceramics, Vol. 68, Nos. 9 – 10, January, 2012 (Russian Original, Nos. 9 – 10, September – October, 2011)

EQUIPMENT

UDC 666.3.022.523

DETERMINATION OF THE STIFFNESS OF AN ELASTIC SUSPENSION OF A VERTICAL VIBRATORY SCREEN AND THE CHARACTERISTIC VIBRATIONAL FREQUENCIES OF THE WORKING ORGAN

V. F. Gaivoronskii[1] and A. I. Postoronko[1, 2]

Translated from *Steklo i Keramika*, No. 10, pp. 32 – 34, October, 2011.

Computational relations for determining the stiffness of an elastic suspension of a vertical vibratory screen taking account factors influencing the efficiency of the screen in filtering ceramic suspensions are presented. The vibrational frequency and amplitude which optimize the operation of the screen while maintaining its service life and the serviceability of its design are recommended.

Key words: spring constant, elastic suspension, dampers, vibrations, imbalance, calculations.

Vibratory setups are used in many areas of industry to sort, unload, and transport industrial materials and for other purposes.

A vertical vibratory screen (see Fig. 1), whose working organ *1* undergoes complicated vibrations in the plane of the screen, was first used in [1] to clean ceramic mixes. We shall assume that the vibrations are linear displacements along the coordinate axes X and Y and rotational motion around the center of gravity (c.g.) of the system. According to experimental data the frequency of the angular motion is

$$\omega = 37 \text{ sec}^{-1},$$

which corresponds to approximately

$$n = \frac{30\omega}{\pi} = \frac{30 \times 37}{3.14} = 353.5 \text{ vibrations/min.} \tag{1}$$

The vibrational frequency of the working organ depends on the mass and stiffness of the elastic suspension (spring) *2*.

Any mechanical structure can be represented as a system of springs, the mass of the vibrating part of the screen, and the dampers *3*. The dampers absorb energy, while the massive bodies and spring do not. Such a system has a resonance at its characteristic frequency. For this reason, as it absorbs energy it starts to vibrate at the characteristic frequency, while the vibrational amplitude depends on the power of the energy source and on the absorption of this energy, i.e., damping inhering in the system itself. In this case the damping is, essentially, a measure of the absorption of vibrational energy, which is created for filtering different ceramic mixes.

Obviously, it is most important to calculate the characteristic frequencies of the linear displacements of the working organ, because they determine the resonance state of the system.

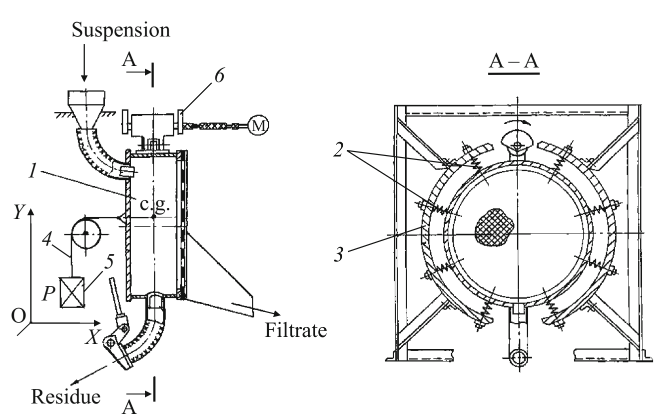

Fig. 1. Vibratory screen.

[1] Ukrainian Engineering-Pedagogical Academy, Slavyansk, Ukraine.
[2] E-mail: slavuipa@mail.ru.

The stiffness of the elastic suspension of the screen is the ratio of the total load on the springs (N) to the magnitude (m) of their deformation.

In the present case the stiffness of the elastic suspension can be determined experimentally. For this purpose a cable *4* is secured to the working organ along the horizontal axis X (see Fig. 1), passing through the center of gravity of the organ, and a weight *5* is suspended to the cable through a block. Knowing the weight P and the deformation l of the elastic suspension by the weight, it is easy to find the stiffness

$$C_{\text{tot}} = P_{\text{av}} / l_{\text{av}}, \tag{2}$$

where C_{tot} is the stiffness of the elastic suspension (total stiffness), N/m; P_{av} is the average load due to the weight, N; and, l_{av} is the average deformation of the elastic suspension, m.

For the construction of the vertical screen it can be assumed to adequate accuracy for practical purposes that

$$C_x = C_y = C_{\text{tot}},$$

where C_X and C_Y are the stiffness of the horizontal and vertical screen, respectively, N/m.

Then the stiffness of one spring is

$$C_{\text{s}} = C_{\text{tot}} / Z, \tag{3}$$

where Z is the number of springs.

The axial stiffness of a spring is (N/m)

$$C_{\text{ax}} = \frac{Ed^4}{8nD^3}, \tag{4}$$

where E is Young's constant with a displacement of the spring material, Pa; d is the diameter of the wire from which a spring is wound, m; D is the average diameter of a spring, m; and, n is the number of working loops of the spring.

The transverse stiffness can be expressed in terms of the axial stiffness

$$C_{\text{trans}} = \frac{C_{\text{ax}}}{1.44\alpha\left[0.204\left(\frac{h_{\text{s}}}{D}\right)^2 + 0.256\right]}, \tag{5}$$

where α is the Rausche coefficient, which depends on the ratio l/h_{s}, $\alpha = 1 - 1.42$, and h_{s} is the working height of the spring, m.

Knowing the stiffness of the elastic suspension and the mass of the working organ together with the imbalance *6*, the characteristic frequency ω_{char} of the vibrations of the working organ can be determined from the expression

$$\omega_{\text{char}}^2 = \frac{C_{\text{tot}}}{M + m}, \tag{6}$$

where ω_{char} is the characteristic vibrational frequency of the working organ, $\sec^{-1}$; M is the mass of the screen parts put into vibration and rigidly tied to the vibrator, kg; and, m is the mass of the imbalances, kg.

Hence it follows that as the spring constant increases the characteristic vibrational frequency of the working organ also increases, while the characteristic vibrational frequency decreases as the mass increases. For a vertical vibratory screen the characteristic vibrational frequencies are approximately 1600 and 2400 vibrations/min (or turns of the imbalance vibrator). Thus, the indicated limits define the resonance region where it is inadmissible to conduct the mass screening process for a given design.

If the system is damped, as all real physical systems are, then the characteristic frequency will be somewhat lower than the value calculated using the relation (6) and will depend on the magnitude of the damping. A 2STS32 stroboscopic tachometer was used for visual monitoring and for studying the vibrational regime when tuning and performing tests on the vibratory screen [1] with such a design and working surface area 0.0415 m^2 (diameter 230 mm) with vibrational frequency 2800 – 3000 vibrations/min of a 25 kg working organ and amplitude 0.7 – 1.5 mm. The amplitude of the vibrations of the working organ was recorded with VR-1, -2 vibrographs. The vibrational frequency of the working organ was monitored with an ATT-6006 universal tachometer.

Testing showed that stable prolonged operation of the screen is possible at vibrational frequency 3000 vibrations/min (60 – 100 Hz). The static moment of the mass of the imbalances was 0.251 N · m. This vibrational regime is optimal, as a result of which the capacity of the screen reached the desired value 6 tons/h when cleaning glaze. It was established that for the given screen design the difference between the displacements at different points of the screen did not exceed 0.1 – 0.6 mm with the indicated vibrational amplitude 0.7 – 1.5 mm.

To adequate accuracy for practical purposes of tuning the vibratory screen the average vibrational amplitude can be determined from the relation

$$A = \frac{K}{M + m}, \tag{7}$$

where A is the amplitude of the vibrations, mm, and K is the static moment of the mass of the imbalances. The value of K can be determined experimentally.

For the given vibratory screen with filtration surface $F = 0.0415$ m^2 the optimal vibrational amplitude is $A = 0.94$ mm.

Many spring – mass – damper systems (i.e., the simplest oscillators that can be used to model the behavior of a mechanical structure) have three degrees of freedom. The vibrational energy of the screen is divided between these degrees of freedom depending on their characteristic frequencies and damping as well as on the frequency of the energy source. For this reason the vibrational energy is never distributed uniformly over the entire screen. And in a vibratory screen

with electric drive the main source of vibrations is the residual imbalance of the motor's rotor. This results in appreciable levels of vibration on the motor's bearings. However, if one of the characteristic frequencies of the screen is close to the vibration at rotor's back vibrational frequency, then vibrations at this frequency can be large even at quite large distances from the motor.

This fact must be taken into account when evaluating the vibrations of a vertical screen cleaning ceramic mixes. In addition, all parts of a structure which has many characteristic frequencies must be followed. At a resonance the vibrational level can become very high and the structure can collapse very quickly.

REFERENCES

1. N. A. Sidorov, V. E. Ved', and A. I. Sidorenko, "Vibratory screen with a vertical screening surface for cleaning ceramic slips," *Steklo Keram.*, No. 11, 25 – 28 (1969); N. A. Sidorov, V. E. Ved', and A. I. Sidorenko, "Vibration screen with vertical screening surface for ceramic slips," *Glass Ceram.*, **26**(11), 677 – 678 (1969).
2. V. F. Gaivoronskii, "Cleaning ceramic suspensions on a vertical vibratory screen," *Naukovi Pratsi Donets'kogo Natsional'nogo Tekhnichnogo Universitetu. Ser. Khimiya Khimich. Tekhnol.*, No. 15(163), 145 – 148 (2010).
3. V. E. Ved' and A. A. Shekhovtsov, "Intensification of the process of cleaning ceramic suspensions," in: *Collection of Works on Processing Suspensions, Emulsions, and Industrial Stagnant Waters* [in Russian], Tekhnika, Kiev (1974), pp. 21 – 25.